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RESEARCH MEMORANDUM

COOLING OF GAS TURBINES

III - ANALYSIS OF ROTOR AND BLADE TEMPERATURES IN LIQUID-COOLED GAS TURBINES

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III - ANALYSIS OF ROTOR AND BLADE TEMPERATURES IN LIQUID-COOLED GAS TURBINES

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SUMMARY

A theoretical analysis of the radial temperature distribution through the rotor and constant cross-sectional area blades near the coolant passages of liquid-cooled gas turbines was made. The analysis was applied to obtain the rotor and blade temperatures of a specific turbine using a gas flow of 55 pounds per second, a coolant flow of 6.42 pounds per second, and an average coolant temperature of 200° F. The effect of using water, ethylene glycol, and kerosene was determined with blades $4\frac{1}{16}$ inches long in which the coolant passages extended to within one-sixteenth inch of the blade tips. The effect of varying blade length was determined for lengths varying from $1\frac{1}{16}$ to $5\frac{1}{16}$ inches, the coolant passages being one-sixteenth inch shorter than the blade length in all cases, with water as the coolant. The effect of varying the coolant-passage lengths from 1 to 4 inches in blades $4\frac{1}{16}$ inches long was also investigated, with water as the coolant. The effective gas temperature was varied from 2000° to 5000° F in each of these investigations. By effective gas temperature is meant the gas temperature that determines the heat flow. Finally, the effect of a variation in the coolant flow was investigated for each of the coolants.

It was found that the blade metal temperatures in the regions near the coolant passages may be kept as low as one-fifth, two-fifths, and one-half the effective gas temperature for water, ethylene glycol, and kerosene coolants, respectively, for a coolant flow of 6.42 pounds per second. The temperature distributions for all blade lengths with coolant passages extending to within one-sixteenth inch of the blade tips were found to show a rotor temperature about equal to the coolant temperature, a steady temperature rise through the rim and a small part of the blade, a nearly constant or prevalent blade temperature through most of the liquid-cooled part of the blade, and a sharp

temperature rise to the blade tip. The results further showed that the coolant passage should be made as long as possible in order to obtain maximum blade cooling. An increase in coolant flow from 2 to 15 pounds per second resulted in doubling the cooling effectiveness. The results of liquid cooling seem to indicate indirectly that large increases in effective gas temperature are possible without the occurrence of metal failures compared to small increases of only 200° F in effective gas temperature obtainable for most blades with rim cooling.

INTRODUCTION

Metals in current use in the construction of turbine wheels and turbine blades limit gas temperatures to about 1500° F at usual speeds; therefore some method of blade cooling is needed if gas temperatures higher than 1500° F are to be used with these metals. The NACA is conducting an investigation of methods of cooling gas turbines that includes indirect blade cooling by removal of heat from the blade roots and tips by conduction and direct blade cooling by the passage of liquid or air through hollow blades. In reference 1, it was shown that some improvement in the blade root-section strength could be achieved by cooling the root and applying a ceramic coating to the blade section near the root, but the effects in cooling the upper half of the blades were negligible unless metals with very large thermal conductivities were used. The effects of the addition of air-cooling fins to a turbine disk were investigated in reference 2. The fins were used in an effort to reduce the rim temperature and hence permit better cooling of the turbine blades. It was found that the blade roots (wheel rims) could be cooled to about 800° F below the effective gas temperature, but that only relatively small gain in cooling was obtained at the critical section of the blade some distance from the blade root. In reference 3, the benefits of rim cooling were demonstrated in terms of turbine operating conditions. For most turbine blades, rim cooling was found to permit 200° F increases in allowable gas temperature.

Lynn, of the Joshua Hendy Iron Works (Sunnyvale, California), proposed liquid cooling using kerosene as the coolant, inasmuch as the kerosene could later be used as fuel. Sanders and Mendelson (reference 4) compared several cooling methods and found possibilities of sufficient improvements in efficiency through air or water cooling to justify extensive research.

Because of the small increases in cooling with rim cooling discussed in references 1 to 3 and of the possibilities of much

larger increases in cooling with liquid cooling suggested in reference 4, a theoretical analysis was made of the radial temperature distribution through the rotor and constant cross-sectional area blades near the coolant passages attainable by passing different liquid coolants through internal passages in a turbine wheel and in constant cross-sectional area turbine blades.

This analysis is given in this report, together with the results of its application to determine the rotor and blade temperatures for a specific turbine using a gas flow of 55 pounds per second, a coolant flow of 6.42 pounds per second, an average coolant temperature of 200° F, effective gas temperatures from 2000° to 5000° F, and water, ethylene glycol, and kerosene as coolants. The effects on the rotor and blade temperatures resulting from (1) varying the blade lengths with coolant passages extending to within one-sixteenth inch of the blade tips, (2) varying the coolant-passage lengths in blades $4\frac{1}{16}$ inches long, and (3) varying the coolant flow rate were investigated.

SYMBOLS

The symbols used in the computations are:

A_1 area of blade tip (section 1), (sq ft)

A_2 cross-sectional area of metal in liquid-cooled section of blade (section 2), (sq ft)

A_3 cross-sectional area of metal in rim (section 3), (sq ft)

A_4 cross-sectional area of constant-strength section of rotor (section 4), (sq ft)

A_t area of blade cooling passage at tip of section 2, (sq ft)

A_r area of metal at tip of section 2, (sq ft)

B number of blades

F prevalent blade temperature, (°F)

K,M,H, J,G,C integration constants

k thermal conductivity of turbine metal, Btu/(hr)(sq ft)(°F/ft)

p_i perimeter of blade-heating surface, (ft)

p_o perimeter of cooling surface, (ft)

q_i heat-transfer coefficient between hot gases and metal, Btu/(hr)(sq ft)($^{\circ}$F)

q_o heat-transfer coefficient between metal and coolant, Btu/(hr)(sq ft)($^{\circ}$F)

q_o' heat-transfer coefficient between metal and cooling air, Btu/(hr)(sq ft)($^{\circ}$F)

r radius, (ft)

$\bar{r}$ average radius, (ft)

T metal temperature, ($^{\circ}$F)

T_a temperature of cooling air, ($^{\circ}$F)

T_e effective temperature of hot gases, ($^{\circ}$F)

T_l average temperature of liquid coolant, ($^{\circ}$F)

x linear coordinate in section 2

y linear coordinate in section 3

z linear coordinate in section 1

$$l = \frac{\text{area of rim}}{\text{area of blades}} = \frac{A_3}{BA_2}$$

$$m = \frac{\text{area of rim exposed to exhaust gas}}{\text{area of metal parts of blades}} = \frac{A_3 - BA_2}{BA_2}$$

$$n = \frac{\text{area of constant-strength section of rotor}}{\text{area of rim}} = \frac{A_4}{A_3}$$

Subscripts:

1 section 1

2 section 2

3 section 3

4 section 4

THEORETICAL ANALYSIS

The equations for the temperature distribution in a turbine wheel and in turbine blades were obtained by equating the heat entering and the heat leaving an element of each of the four sections into which the turbine wheel and blades were divided (fig. 1). In the derivation of these equations, the following assumptions were made:

1. The values of A, k, p_i, p_o, q_i, q_o, T_e, and T_l were constant for any given section. It must be noted that the coolant temperature in the inlet passage was less than that in the outlet passage. The increase in the temperature in the one passage compensated for the decrease in the temperature in the other passage, so that an average temperature T_l for any blade cross section could be considered constant.

2. The values of q_i and T_e over the blade tip were the same as over the blade.

3. Heat flows were equated at the junction of the various sections.

4. No temperature gradients other than radial were considered.

5. About 10 percent of the value of q_i was attributed to radiation.

6. Cooling air, in contact with both sides of the rim and rotor, was assumed to remove heat.

The derivations of the temperature-distribution equations are as follows:

Part of blade not liquid-cooled (section 1). -

$$\text{Heat entering from top} = -kA_1 \frac{dT}{dz}$$

$$\text{Heat entering from sides} = p_i q_i (T_e - T)\, dz$$

$$\text{Heat leaving bottom} = -kA_1 \frac{dT}{dz} - kA_1 \frac{d^2T}{dz^2} dz$$

Therefore

$$-kA_1 \frac{dT}{dz} + p_i q_i \ (T_e - T) \ dz = -kA_1 \frac{dT}{dz} - kA_1 \frac{d^2T}{dz^2} dz$$

or

$$\frac{d^2T}{dz^2} - \nu^2 T = -\nu^2 T_e$$

where

$$\nu^2 = \frac{p_i q_i}{kA_1}$$

A solution is

$$T = T_e - C \cosh \nu(z + \lambda)$$

where C and λ are integration constants and λ is evaluated by applying the terminal condition at the blade tip; that is,

$$-kA_1 \frac{dT}{dz} = q_i A_1 \ (T_e - T)$$

or

$$\tanh \nu(z_1 + \lambda) = \frac{q_i}{k\nu} = \sqrt{\frac{q_i A_1}{p_i k}}$$

<u>Part of blade liquid-cooled (section 2)</u>. -

$$\text{Heat entering from top} = -kA_2 \frac{dT}{dx}$$

$$\text{Heat entering from sides} = p_i q_i \ (T_e - T) \ dx$$

$$\text{Heat leaving bottom} = -kA_2 \frac{dT}{dx} - kA_2 \frac{d^2T}{dx^2} dx$$

$$\text{Heat entering coolant} = p_{o,2} \ q_{o,2} \ (T - T_l) \ dx$$

Therefore

$$-kA_2 \frac{dT}{dx} + p_i q_i \; (T_e - T) \; dx = -kA_2 \frac{dT}{dx} - kA_2 \frac{d^2T}{dx^2} dx + p_{o,2} \; q_{o,2} \; (T - T_l) \; dx$$

or

$$\frac{d^2T}{dx^2} - \mu^2 T = -\gamma^2$$

where

$$\mu^2 = \frac{p_i q_i + p_{o,2} \, q_{o,2}}{kA_2}$$

$$\gamma^2 = \frac{p_i q_i \, T_e + p_{o,2} \, q_{o,2} \, T_l}{kA_2}$$

A solution is

$$T = F - He^{\mu x} - Je^{-\mu x}$$

where

$$F = \frac{p_i q_i \, T_e + p_{o,2} \, q_{o,2} \, T_l}{p_i q_i + p_{o,2} \, q_{o,2}}$$

and H and J are integration constants.

Rotor rim (section 3). - A simple solution for the temperature distribution in the rotor rim was obtained by disregarding curvature, which is justified because the rim thickness is small in comparison with the rim radius. An average value for $p_{o,3} \, q_{o,3}$ was used.

The heat-balance equation was found to be

$$\frac{d^2T}{dy^2} - \beta^2 T = -\delta^2$$

where

$$\beta^2 = \frac{4\pi \bar{r}_3 q_o' + p_{o,3} \, q_{o,3}}{kA_3}$$

and

$$\delta^2 = \frac{4\pi \bar{r}_3 q_o' \, T_a + p_{o,3} \, q_{o,3} \, T_l}{kA_3}$$

A solution is

$$T = L + Ke^{\beta y} + Me^{-\beta y}$$

where

$$L = \frac{4\pi \bar{r}_3 q_o' \, T_a + p_{o,3} \, q_{o,3} \, T_l}{4\pi \bar{r}_3 q_o' + p_{o,3} \, q_{o,3}}$$

and K and M are integration constants.

Constant-strength section of rotor (section 4). - In reference 2 an exact solution for the temperature distribution through the turbine rotor was found. Accuracy was needed, because most of the heat was carried off by the rotor (a temperature drop of 450° F occurred in the rotor). In the present case, however, most of the heat is removed by the liquid coolant (the temperature drop through the rotor is only a few degrees), and the rotor temperatures are sufficiently low to cause no cooling problems. Consequently, an approximate solution for the temperature distribution through the rotor was obtained in the present investigation by use of an average radius and an average area. The heat-balance equation was found to be

$$\frac{d^2T}{dr^2} - \alpha^2 T = -\zeta^2$$

where

$$\alpha^2 = \frac{4\pi \bar{r}_4 q_o' + p_{o,4} \, q_{o,4}}{kA_4}$$

and

$$\zeta^2 = \frac{4\pi \bar{r}_4 q_o' \, T_a + p_{o,4} \, q_{o,4} \, T_l}{kA_4}$$

A solution is

$$T = N + G \cosh \alpha r$$

where

$$N = \frac{4\pi \bar{r}_4 q_o' \, T_a + \theta_{o,4} \, q_{o,4} \, T_l}{4\pi \bar{r}_4 q_o' + p_{o,4} \, q_{o,4}}$$

and G is an integration constant. The boundary condition $dT/dr = 0$ for $r = 0$ has been applied.

The values of the six integration constants were found by solving simultaneously the six equations resulting from equating the temperatures and the heat flows at the various junction points. At the junction of sections 1 and 2 ($z = z_2$ and $x = x_1$)

$$T_e - C \cosh \nu(z_2 + \lambda) = F - He^{\mu x_1} - Je^{-\mu x_1}$$

$$kA_1 C \sinh \nu(z_2 + \lambda) = kA_r \mu (He^{\mu x_1} - Je^{-\mu x_1}) + q_o A_t \left(F - He^{\mu x_1} - Je^{-\mu x_1} - T_l\right) \tag{1}$$

Equation (1) equates the heat leaving section 1 to the sum of the heat entering the metal of section 2 and the heat entering the coolant at that part of the blade where the inlet and the outlet passages are connected. The area of the metal at this blade cross section is denoted by A_r. The use of A_r implies an approximation in the procedure; that is, a separate section for that part of the blade containing the passage that connects the inlet and the outlet passages has not been introduced. An investigation showed that the use of such a section would slightly decrease the temperatures at the blade tip.

At the junction of sections 2 and 3 ($x = x_2$ and $y = y_1$)

$$F - He^{\mu x_2} - Je^{-\mu x_2} = L + Ke^{\beta y_1} + Me^{-\beta y_1}$$

$$kA_2 \mu \left(He^{\mu x_2} - Je^{-\mu x_2}\right) + mq_i A_2 \left(F - He^{\mu x_2} - Je^{-\mu x_2} - T_l\right) = \beta k A_2 l \left(Me^{-\beta y_1} - Ke^{\beta y_1}\right) \tag{2}$$

Equation (2) equates the sum of the heat leaving section 2 and that entering section 3 directly from the hot gases to the total heat entering section 3.

At the junction of sections 3 and 4 ($y = y_2$ and $r = r_1$)

$$L + Ke^{\beta y_2} + Me^{-\beta y_2} = N + G \cosh \alpha r_1$$

$$k\beta \left(-Ke^{\beta y_2} + Me^{-\beta y_2}\right) = nk\alpha G \sinh \alpha r_1$$

APPLICATION OF ANALYSIS

The following assumptions were made in applying the previous results to specific numerical calculations:

1. An average thermal conductivity k of 15 Btu per hour per square foot per °F per foot was used.

2. The gas flow was 55 pounds per second. The heat-transfer coefficient q_i corresponding to this gas flow was calculated by the method of reference 5, page 236 and found to be 217 Btu per hour per square foot per °F. Because rim cooling was not expected to be adequate, small turbines were chosen for the rim-cooling investigations. The indications are, however, that liquid cooling will be very adequate, and a more powerful turbine with high gas flow was chosen for the liquid-cooling studies. The use of a high heat-transfer coefficient tends to minimize the advantages of liquid cooling when comparisons are made with results obtained for rim cooling in references 2 and 3 because these reports used a low heat-transfer coefficient.

3. The following passages were assumed: (a) two 1/4-inch-diameter passages in each blade, (b) two 1/4-inch-diameter passages running circumferentially through the rim, and (c) ten 1/2-inch-diameter passages running radially through the constant-strength section of the rotor, connected to feed passages near the axis.

4. The liquid coolants were considered at average temperatures of 200° F and a flow of 6.42 pounds per second. This flow rate assures turbulent flow in all cases, making possible a better comparison of liquids. The corresponding heat-transfer coefficients were calculated (reference 5, p. 168) as

$$q_{o,2} = 2370 \text{ Btu/(hr)(sq ft)(°F)} \qquad \text{(water)}$$

$$q_{o,2} = 649 \text{ Btu/(hr)(sq ft)(°F)} \qquad \text{(ethylene glycol)}$$

$$q_{o,2} = 510 \text{ Btu/(hr)(sq ft)(°F)} \qquad \text{(kerosene)}$$

5. By use of the assumed values of coolant flow, gas flow, and passage dimensions, estimates were obtained (reference 5, p. 168) for the values of $p_o q_o$ for the rim and the rotor and these values were assumed to be uniformly distributed throughout the respective sections. They were

$p_{o,3}\,q_{o,3} = 30{,}360$ Btu/(hr)(ft)(^{o}F) (water)

$p_{o,3}\,q_{o,3} = 8305$ Btu/(hr)(ft)(^{o}F) (ethylene glycol)

$p_{o,3}\,q_{o,3} = 6533$ Btu/(hr)(ft)(^{o}F) (kerosene)

$p_{o,4}\,q_{o,4} = 6107$ Btu/(hr)(ft)(^{o}F) (water)

$p_{o,4}\,q_{o,4} = 1656$ Btu/(hr)(ft)(^{o}F) (ethylene glycol)

$p_{o,4}\,q_{o,4} = 1318$ Btu/(hr)(ft)(^{o}F) (kerosene)

6. Air at 0^{o} F cooled the rim and the rotor and the heat-transfer coefficient q_o' was 30 Btu per hour per square foot per ^{o}F.

7. The effective temperature of the hot gases T_e was regarded as the gas temperature at the blades. (In reference 3 the relation between the effective gas temperature and the inlet gas temperature is given in detail.)

8. The turbine considered had 55 blades.

9. Some numerical values used were

$$A_1 = 0.00198 \text{ square foot}$$

$$A_3 = 0.312 \text{ square foot}$$

$$A_4 = 0.236 \text{ square foot}$$

$$p_{o,2} = 0.131 \text{ foot}$$

$$p_1 = 0.25417 \text{ foot}$$

$$\bar{r}_3 = 0.4917 \text{ foot}$$

$$\bar{r}_4 = 0.3333 \text{ foot}$$

RESULTS AND DISCUSSION

The specific conditions for which temperature distributions in a gas turbine were determined are given in the following table:

Blade length (in.)	Length of cooling passage (in.)	Coolant	Effective gas temperature (°F)
$5\frac{1}{16}$	5	Water	2000-5000
$4\frac{1}{16}$	4	Water	2000-5000
$4\frac{1}{16}$	4	Ethylene glycol	2000-5000
$4\frac{1}{16}$	4	Kerosene	2000-5000
$4\frac{1}{16}$	3	Water	2000-5000
$4\frac{1}{16}$	2	Water	2000-5000
$4\frac{1}{16}$	1	Water	2000-5000
$3\frac{1}{16}$	3	Water	2000-5000
$2\frac{1}{16}$	2	Water	2000-5000
$1\frac{1}{16}$	1	Water	2000-5000

The large flow of heat near the cooling fluid may produce considerable temperature differences between parts of the metal near the coolant and at some distance from the coolant. Inasmuch as the inside heat-transfer coefficient is so much larger than the outside heat-transfer coefficient, the metal temperatures obtained in this analysis would be expected to be a close approach to the values near the coolant. No heat flow through the blade cross section was considered in this analysis.

Each of the temperature-distribution curves shows a nearly constant rotor temperature in the neighborhood of 200° F, a steady temperature rise through the rim and a small part of the blade, a nearly constant temperature through most of the liquid-cooled part of the blade, and a sharp temperature rise to the blade tip. A more careful examination of the curves indicates that the prevalent blade temperature F is about one-fifth the effective gas temperature when the turbine is cooled by water, about two-fifths when cooled by ethylene glycol, and about one-half when cooled by kerosene.

Figure 2 presents temperature distributions for turbines with blades of different lengths cooled by water to within one-sixteenth inch of the blade tips. The increase in temperature over the prevalent blade temperature is the same in each case but this change takes place nearer the blade tips for the longer blades; the change occurs in the last one-half inch for a blade $4\frac{1}{16}$ inches long.

The temperature distributions for turbines with blades $4\frac{1}{16}$ inches long, cooled by water 1, 2, 3, and 4 inches along the blades are given in figure 3. The shorter the blade cooling passages, the less effective is the cooling. The temperature rises through the part of the blades that is not liquid-cooled until the blade temperature approaches the effective gas temperature at the blade tips.

Figure 4 shows temperature distributions for turbines with blades $4\frac{1}{16}$ inches long cooled 4 inches along the blades by water, ethylene glycol, and kerosene plotted on the basis of effective gas temperature. Water is by far the most efficient of the three coolants.

The effect on the prevalent blade temperature F of varying the coolant flow is shown in figure 5. For a coolant flow of 6.42 pounds per second, the prevalent blade temperature is about one-fifth, two-fifths, or one-half the effective gas temperature for water, ethylene glycol, and kerosene coolants, respectively. A coolant flow of 15 pounds per second results in cooling twice as effective as a coolant flow of 2 pounds per second. It must be remembered that better cooling can be obtained by increasing the cooling surfaces.

CONCLUSIONS

From a theoretical analysis of radial blade temperatures near the coolant passage in liquid-cooled gas turbines and for the conditions used in this investigation, the following conclusions are drawn:

1. For a coolant flow of 6.42 pounds per second, the blade metal temperatures in the regions near the coolant passages may be kept as low as one-fifth, two-fifths, and one-half the effective gas temperature for water, ethylene glycol, and kerosene coolants, respectively.

2. The temperature distributions for all blade lengths with coolant passage extending to within one-sixteenth inch of the blade tips were found to show a rotor temperature about equal to the coolant

temperature, a steady temperature rise through the rim and a small part of the blades, a nearly constant or prevalent blade temperature through most of the liquid-cooled part of the blades, and a sharp temperature rise to the blade tips.

3. An increase in the length of the coolant passages in a $4\frac{1}{16}$ inch blade results in extending the prevalent blade temperature over a greater blade distance; i.e., the longer the coolant passages, the longer is that portion of the blade whose temperature is about one-fifth the effective gas temperature if water at a flow of 6.42 pounds per second is used as the coolant.

4. An increase in coolant flow from 2 to 15 pounds per second results in doubling the cooling effectiveness in the turbine blades.

5. The results of liquid cooling indirectly indicate that large increases in effective gas temperature are possible without the occurrence of metal failures compared to the small increase of only 200° F in effective gas temperature obtainable for most blades with rim cooling.

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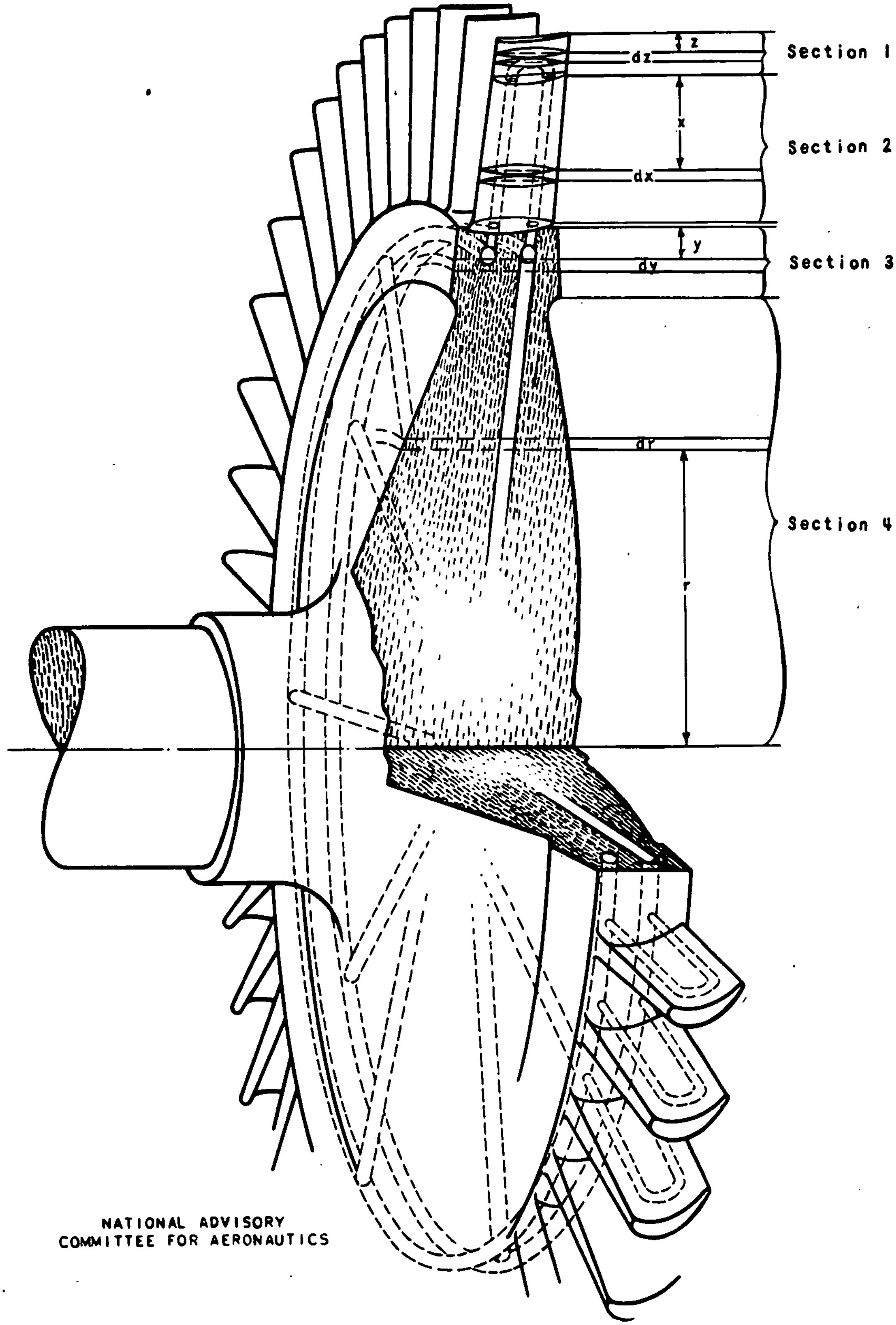


Figure 1. - Arrangement of internal-cooling passages.

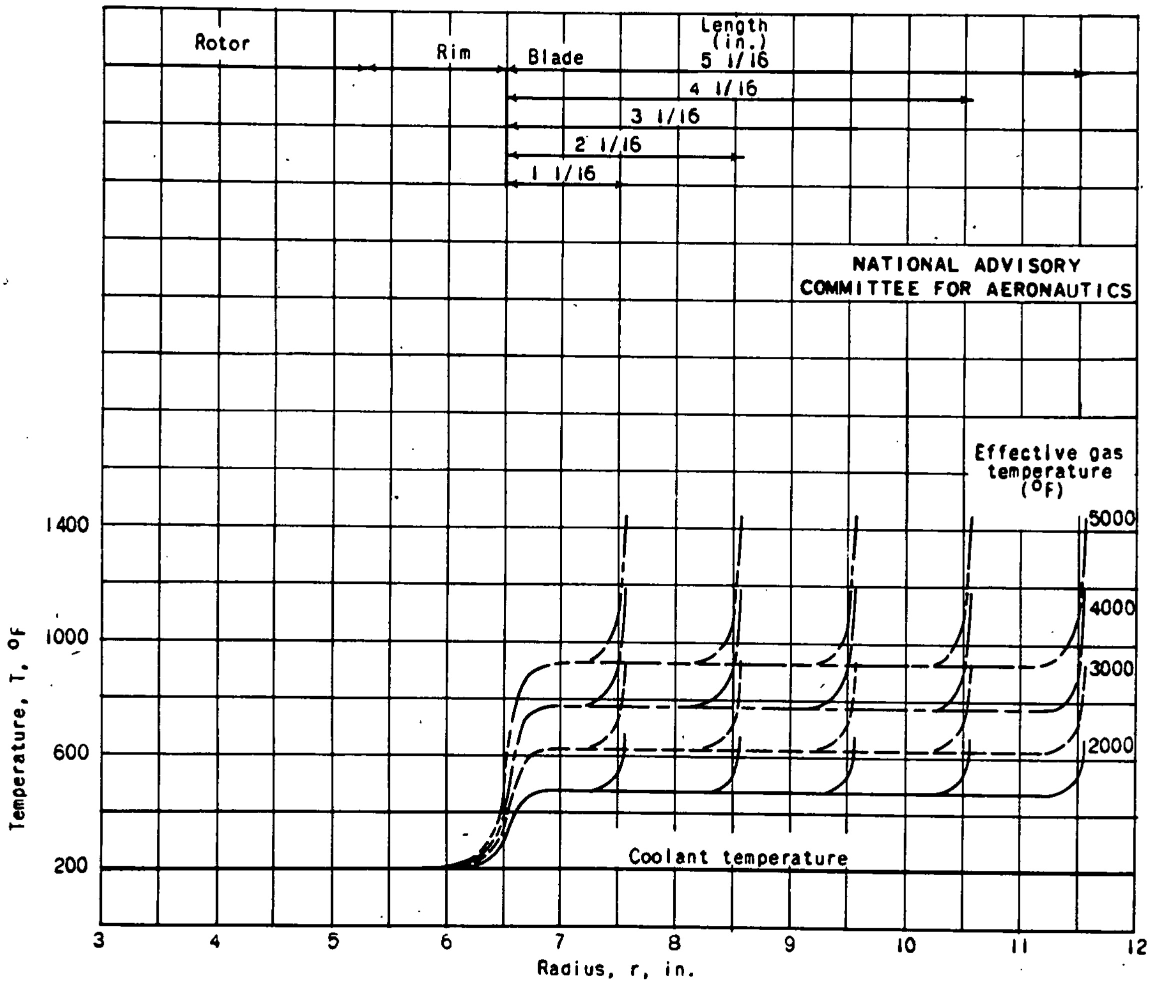
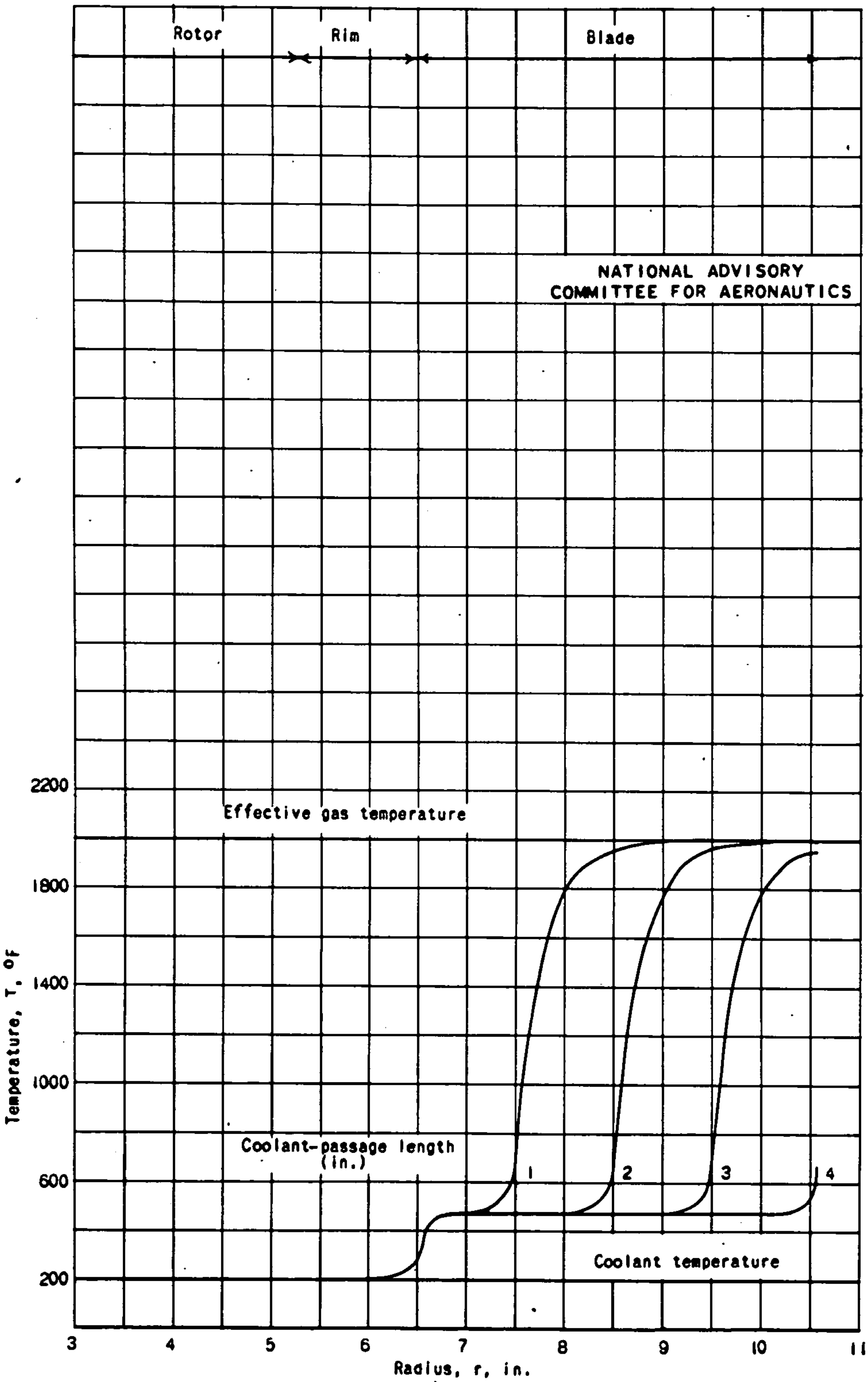
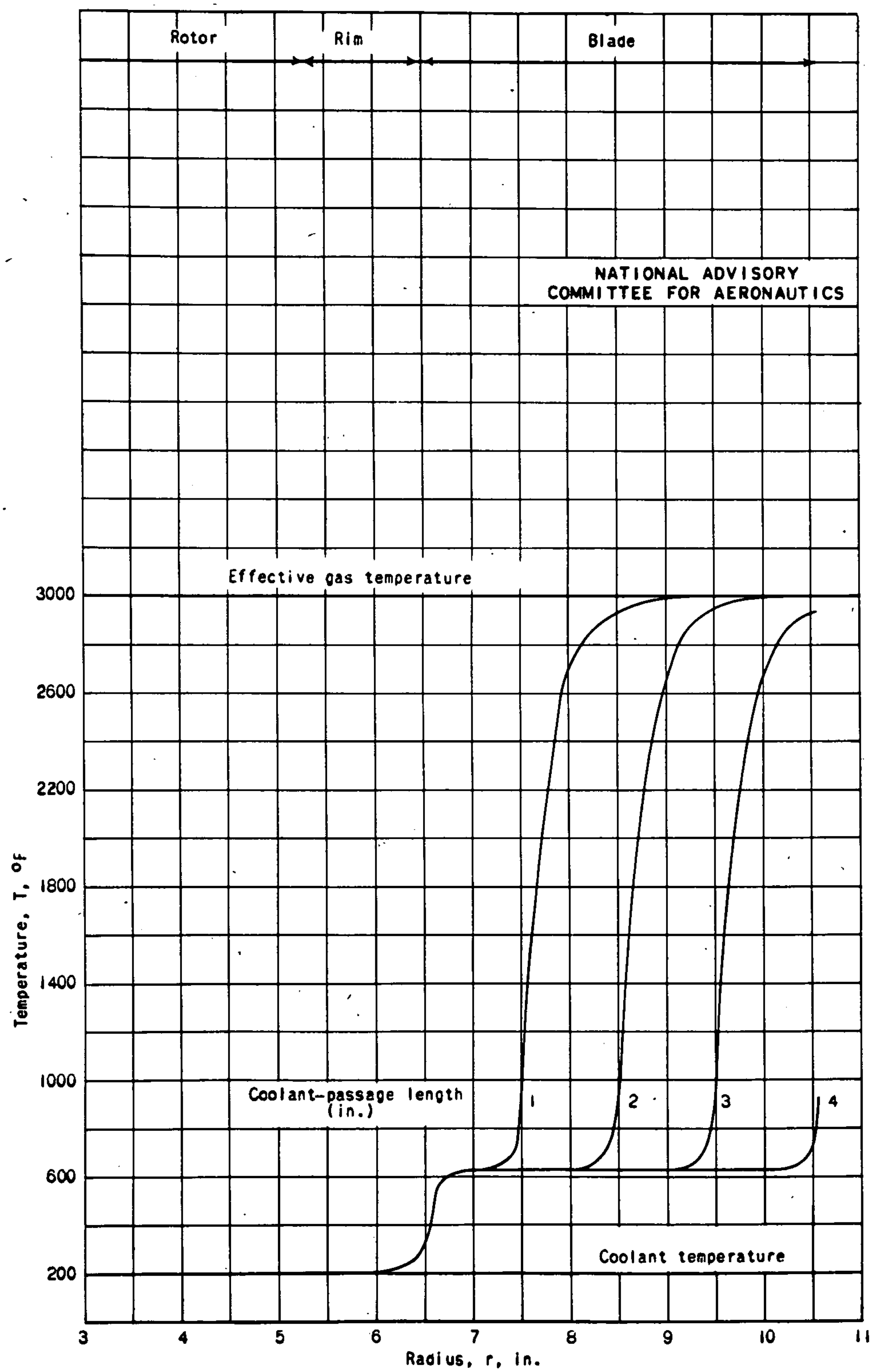


Figure 2. - Effect of various blade lengths on temperature distribution for gas turbine cooled by water at 200° F for effective gas temperatures from 2000° to 5000° F. Coolant passages extend to within one-sixteenth inch of blade tip.



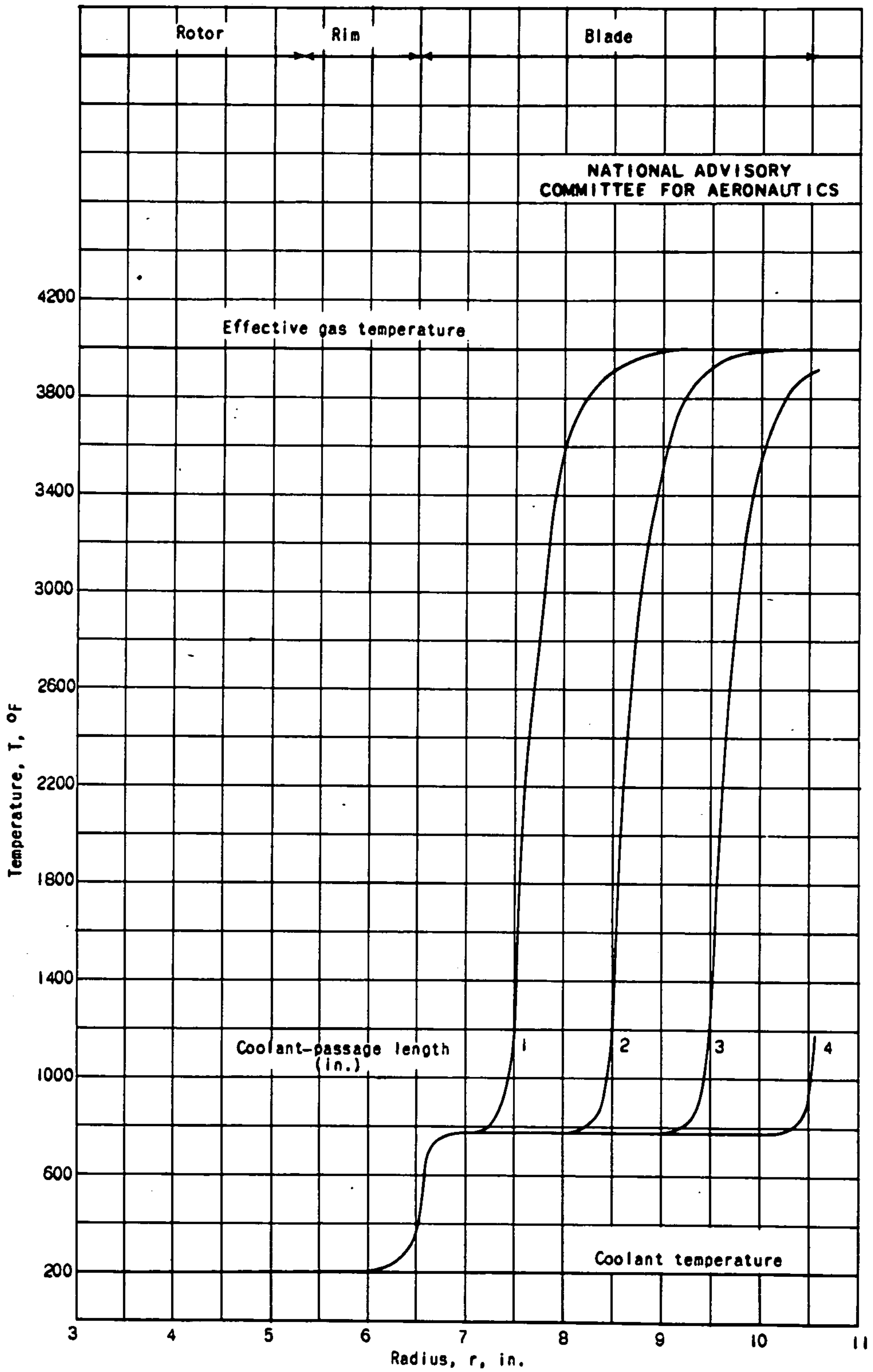
(a) Effective gas temperature, T_e, 2000° F.

Figure 3. - Effect of various coolant-passage lengths on temperature distribution for a gas turbine cooled by water at 200° F for effective gas temperature of 2000° F. Blade length, 4 1/16 inch.



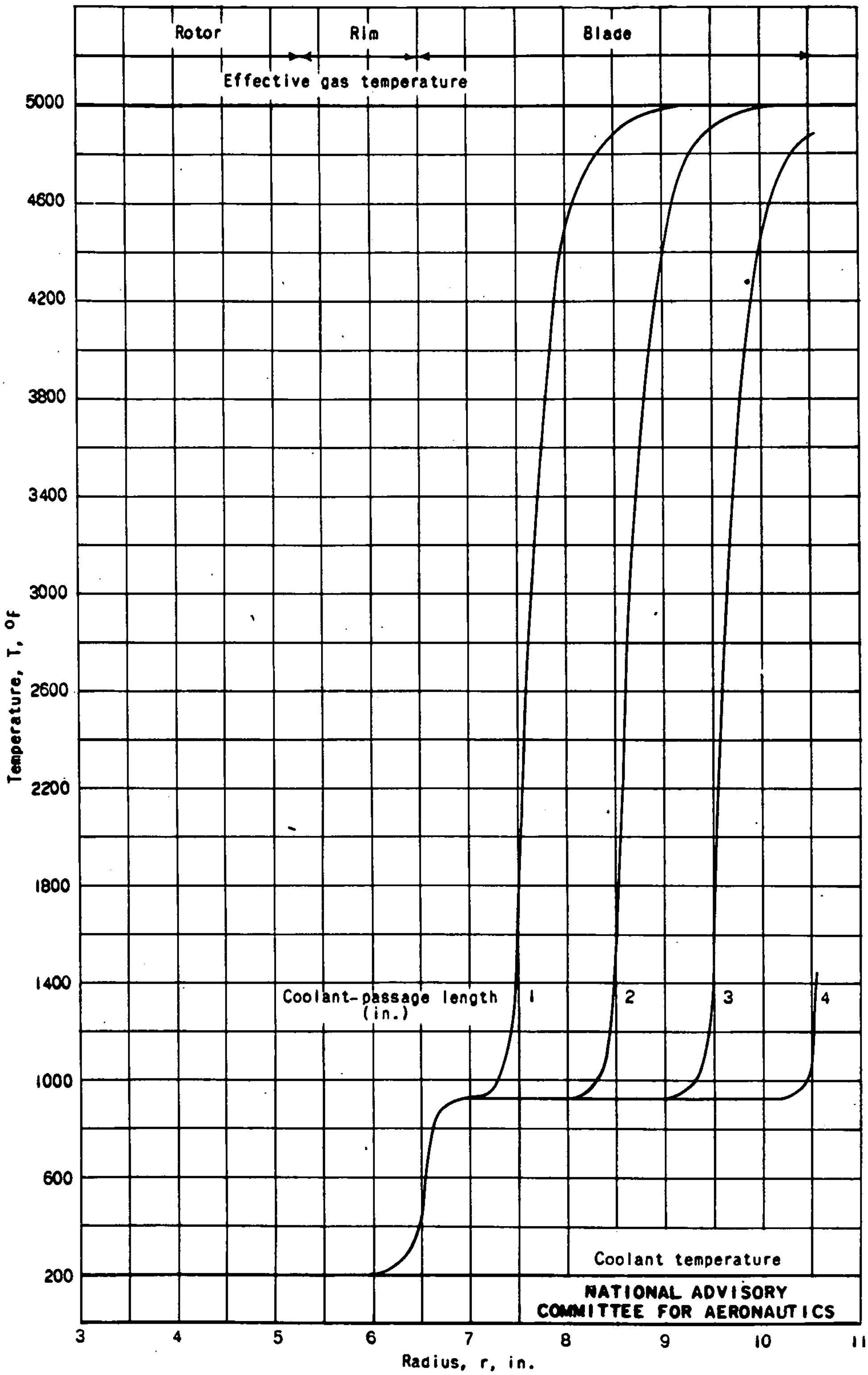
(b) Effective gas temperature, T_e, 3000° F.

Figure 3. - Continued. Effect of various coolant-passage lengths on temperature distribution for a gas turbine cooled by water at 200° F for effective gas temperature of 3000° F. Blade length, 4 1/16 inch.



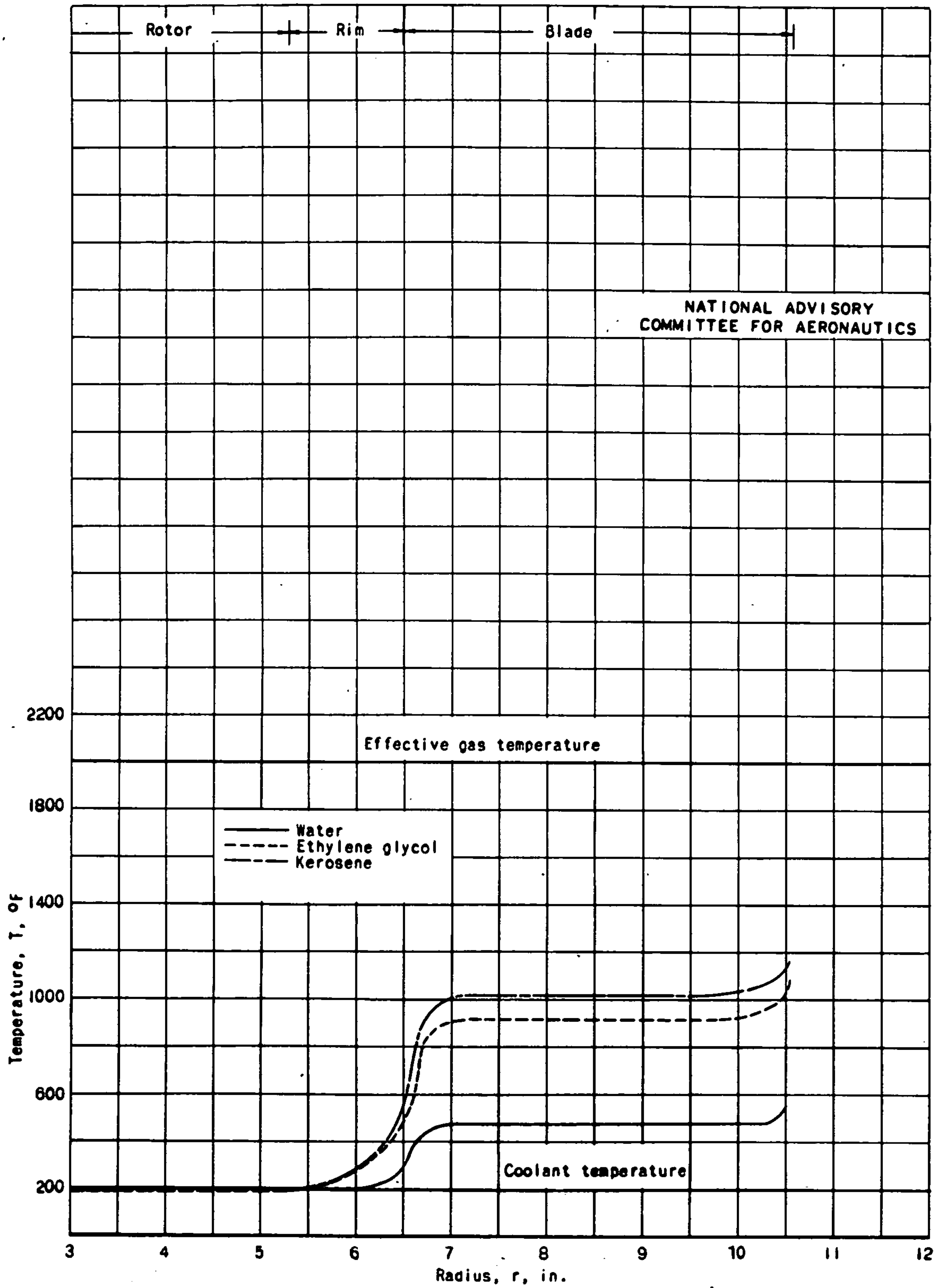
(c) Effective gas temperature, T_e, 4000° F.

Figure 3. - Continued. Effect of various coolant-passage lengths on temperature distribution for a gas turbine cooled by water at 200° F for effective gas temperature of 4000° F. Blade length, 4 1/16 inch.



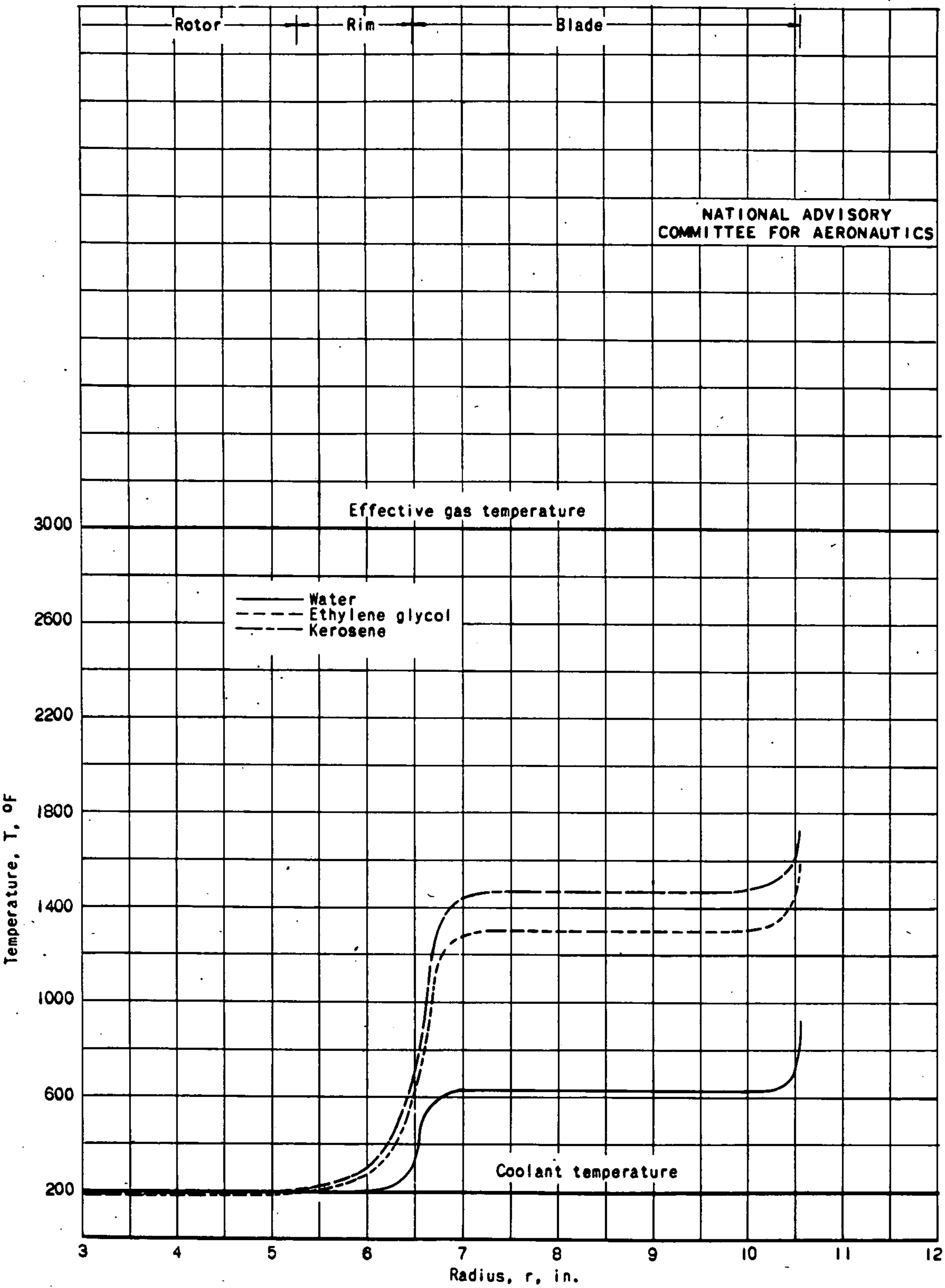
(d) Effective gas temperature, T_e, 5000° F.

Figure 3. - Concluded. Effect of various coolant-passage lengths on temperature distribution for a gas turbine cooled by water at 200° F for effective gas temperature of 5000° F. Blade length, 4 1/16 inch.



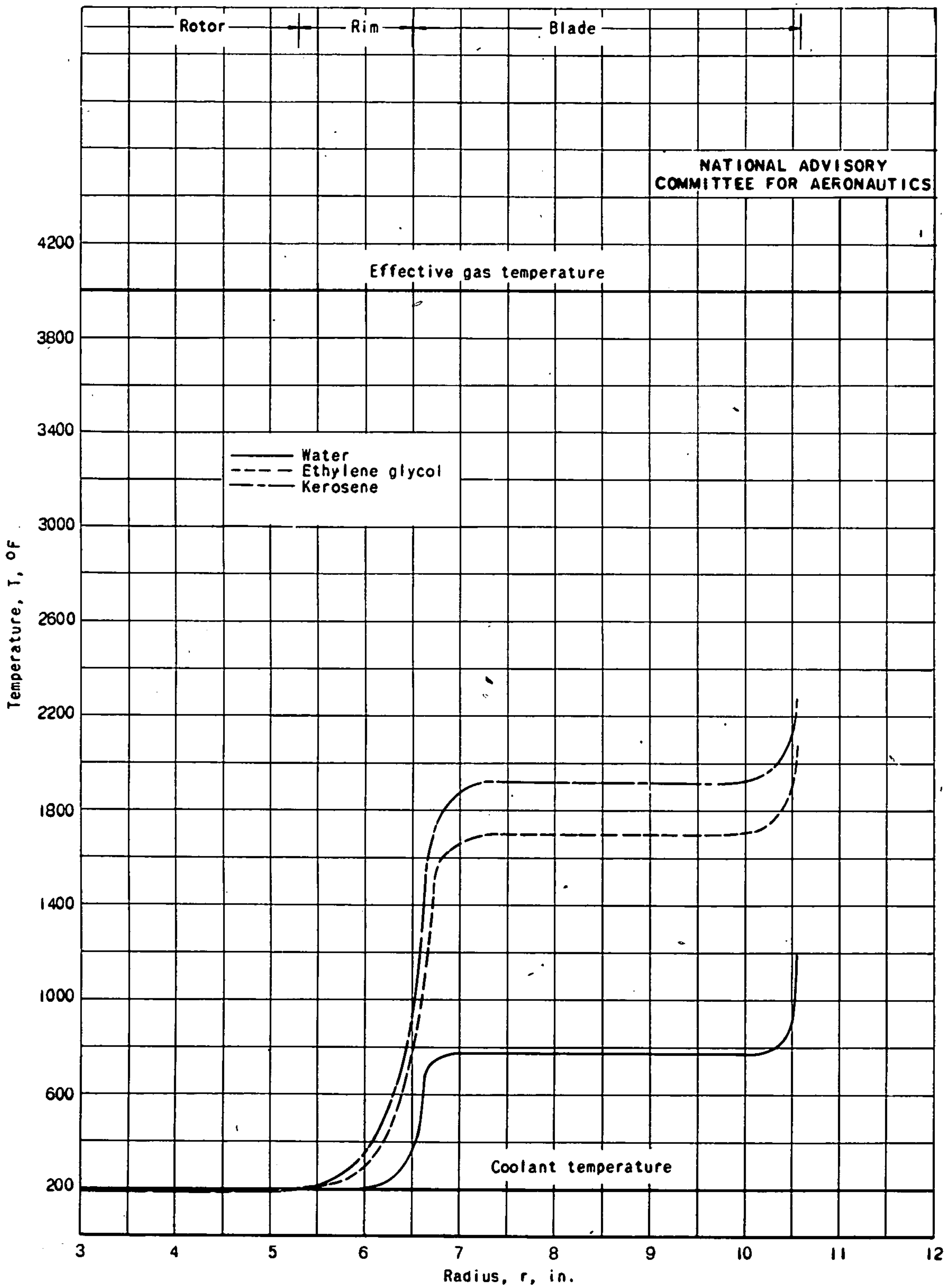
(a) Effective gas temperature, T_e, 2000° F.

Figure 4. - Temperature distribution for a gas turbine for various liquid coolants at 200° F; coolant passages are 4 inches long and extend to within one-sixteenth inch of blade tip.



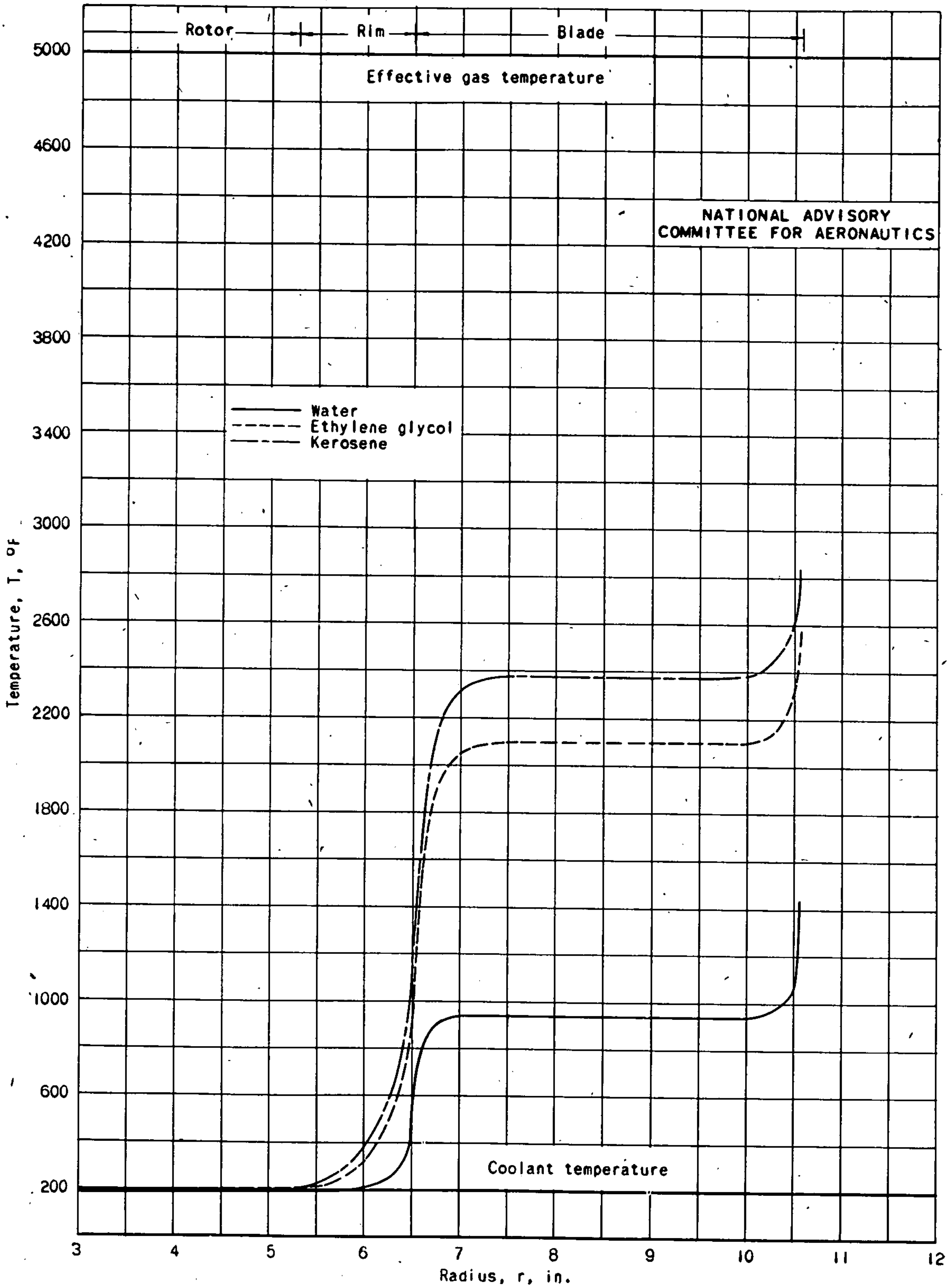
(b) Effective gas temperature, T_e, 3000° F.

Figure 4. - Continued. Temperature distribution for a gas turbine for various liquid coolants at 200° F; coolant passages are 4 inches long and extend to within one-sixteenth inch of blade tip.



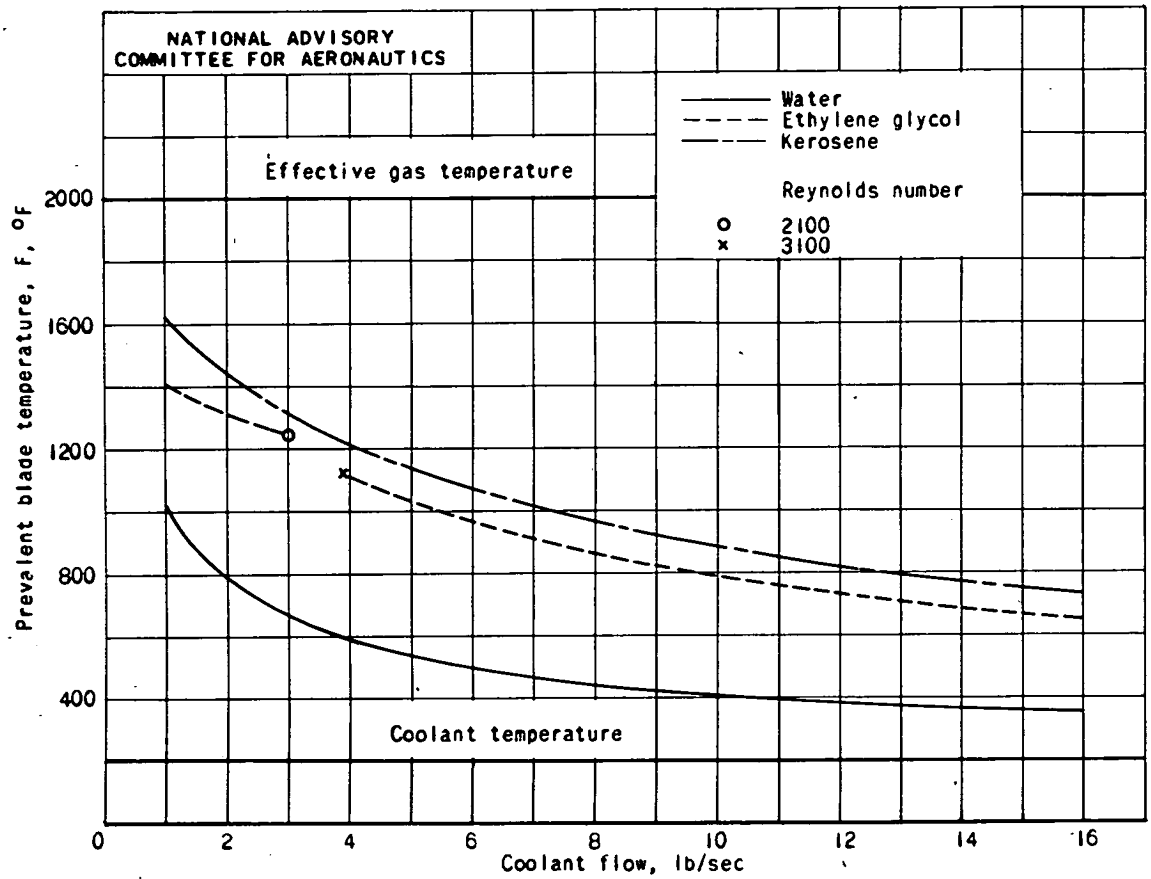
(c) Effective gas temperature, T_e, 4000° F.

Figure 4. - Continued. Temperature distribution for a gas turbine for various liquid coolants at 200° F; coolant passages are 4 inches long and extend to within one-sixteenth inch of blade tip.



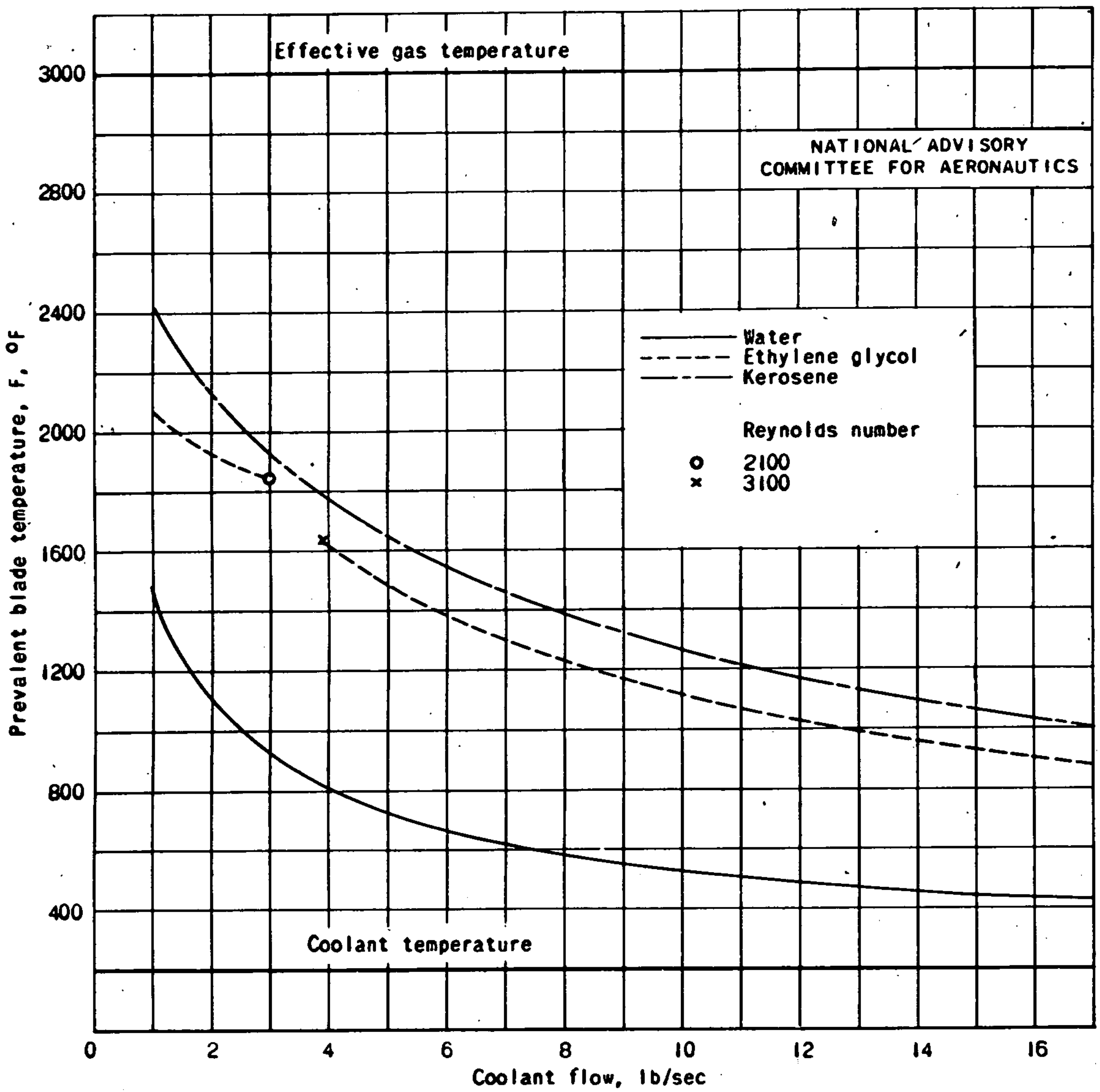
(d) Effective gas temperature, T_e, 5000° F.

Figure 4. - Concluded. Temperature distribution for a gas turbine for various liquid coolants at 200° F; coolant passages are 4 inches long and extend to within one-sixteenth inch of blade tip.



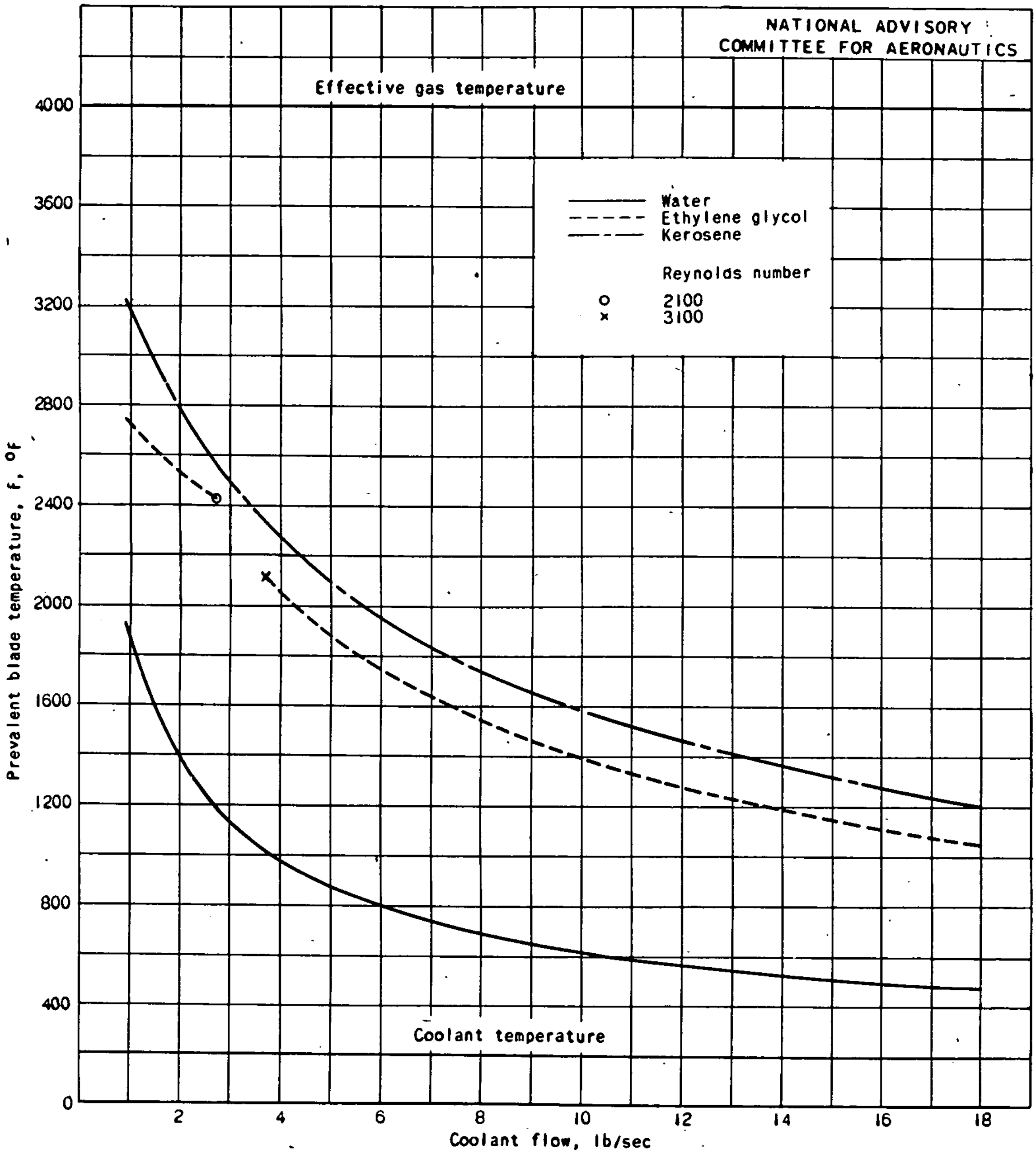
(a) Effective gas temperature, T_e, 2000° F

Figure 5. - Variation of prevalent blade temperature with coolant flow.



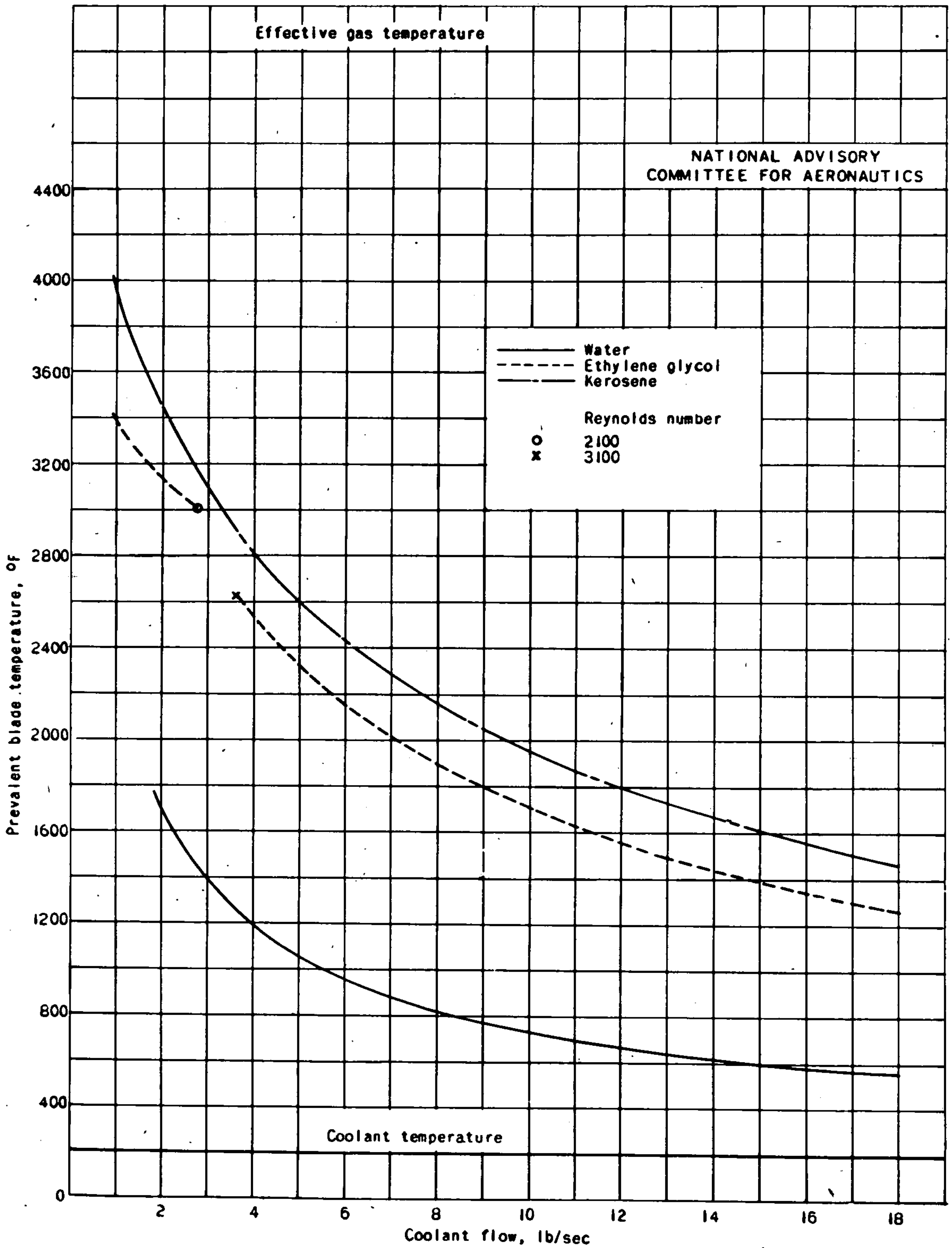
(b) Effective gas temperature, T_e, 3000° F.

Figure 5. - Continued. Variation of prevalent blade temperature with coolant flow.



(c) Effective gas temperature, T_e, 4000° F.

Figure 5. - Continued. Variation of prevalent blade temperature with coolant flow.



(d) Effective gas temperature, T_e, 5000° F.

Figure 5. - Concluded. Variation of prevalent blade temperature with coolant flow.